

## Our rotor modeling service defines a new balancing standard for a repaired turbine rotor



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**B**ently Nevada provides hardware and software products and services to help customers manage their machinery. Software products, such as ADRE® for Windows® and Data Manager® 2000, can be used to capture vibration and process data during startups, shutdowns, and steady state conditions. Startup and shutdown databases can provide information that defines the frequency of balance resonances and direct and quadrature stiffness properties of a rotor system.

We recently added a service to further help customers manage their machinery. Our Machinery Management Services Engineers can model a rotor and evaluate its response to perturbation forces, such as unbalance. One of the unique capabilities of the program is its ability to integrate startup and shutdown databases into the development of models. These databases represent the actual behavior of the machine under field conditions. The rotor model can be adjusted until the computer-calculated response matches the observed behavior from the field data. Once the model has been verified, it can be used to predict the behavior of the rotor under various forcing functions. The verification process yields a high level of accuracy and confidence in the rotor model. This capability is unique to our industry.

A combined cycle power plant recently experienced two incidences of major rub malfunctions on their steam turbine. The first rotor rub occurred when a turbine blade broke off the rotor during full-power operation. The resulting high unbalance force caused a high level of deflection in

the midspan area of the rotor and caused a rotor rub.

Subsequent inspection of the rotor revealed extensive rotor and seal damage and additional turbine blading damage. The rotor was removed and sent to a repair facility for refurbishment and a high-speed balance.

After the rotor was returned to the power plant and installed, an attempt was made to recommission the turbine. During the initial startup, plant personnel followed their normal startup and heat-soaking procedures. They accelerated the turbine to 2080 rpm and then held the rotor speed constant to observe the vibration levels. The 2080 rpm speed is just below the frequency of the first balance resonance. The vibration levels at each turbine bearing were approximately  $35.6 \mu\text{m}$  (1.4 mil) pp. This was considered acceptable to permit acceleration through the balance resonance region. However, as they attempted to accelerate through this region, the rotor midspan again made contact with the seals in another major rub event. The turbine was shut down immediately, and the rotor and seals were inspected.

Significant damage to the rotor and seals had occurred, which necessitated their removal and repair. Since plant personnel had followed their normal startup procedures, they were puzzled why this unit experienced this rub. During many successful startups, this turbine had never had rub problems. They were equally puzzled by the magnitude of the damage, since the vibration levels at the bearings were considered acceptable.

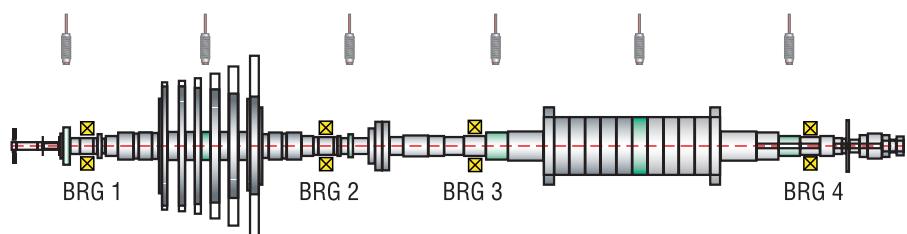


Figure 1. Computer model of steam turbine generator rotor.

## Results of the rotor modeling program

Bently Nevada's Machinery Management Services organization was contacted to help explain the observed behavior of the machine. Drawings were obtained, a rotor model was created (Figure 1), and the computer-generated forced response provided the modal information to define the midspan deflection of the turbine rotor. Discussions were also held on the nature of the rotor repairs after the initial rub event. The damaged sections of the rotor required remachining to smaller shaft diameters. The direct stiffness property of the rotor had been reduced, which allowed the rotor to have higher midspan deflections in response to residual unbalance forces.

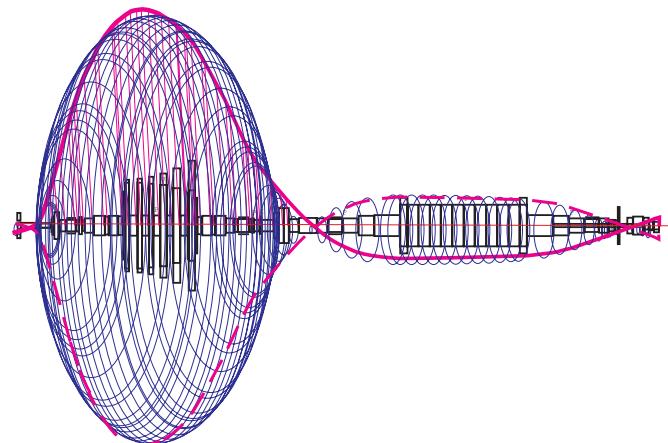
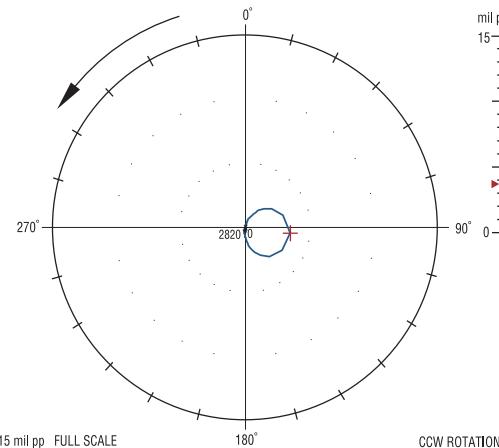


Figure 2. Mode shape of modified rotor at balance resonance frequency.

POINT: TURB\_GOV\_END\_BRG\_VERT 0° 1X 3.68∠96° @2160 RPM  
From 13Sep99 16:26:07 To 13Sep99 16:26:07 TRANSIENT



POINT: TURB\_MID\_SPAN\_BRG\_VERT 0° 1X 21.7∠89° @2160 RPM  
From 13Sep99 16:26:02 To 13Sep99 16:26:02 TRANSIENT

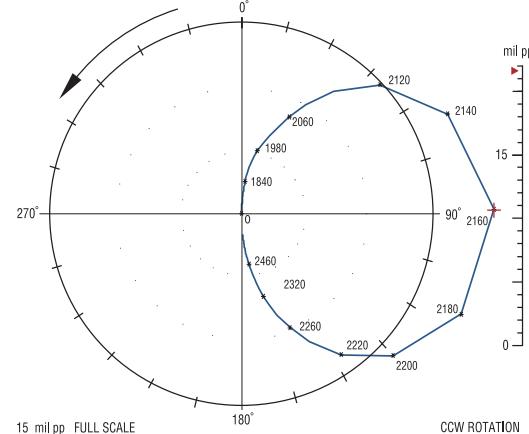
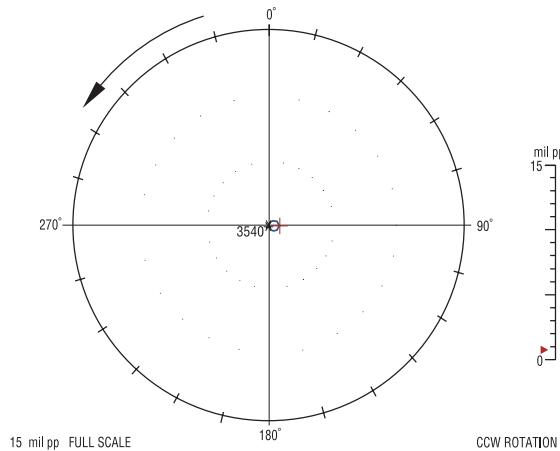


Figure 3. Polar plots showing the calculated response of modified turbine to original residual unbalance standard at Bearing #1 and turbine midspan.

POINT: TURB\_GOV\_END\_VERT 0° 1X 0.810∠95° @2160 RPM  
From 07Oct99 11:20:48 To 07Oct99 11:20:48 TRANSIENT



POINT: TURB\_MID\_SPAN\_VERT 0° 1X 4.78∠90° @2160 RPM  
From 07Oct99 11:20:48 To 07Oct99 11:20:48 TRANSIENT

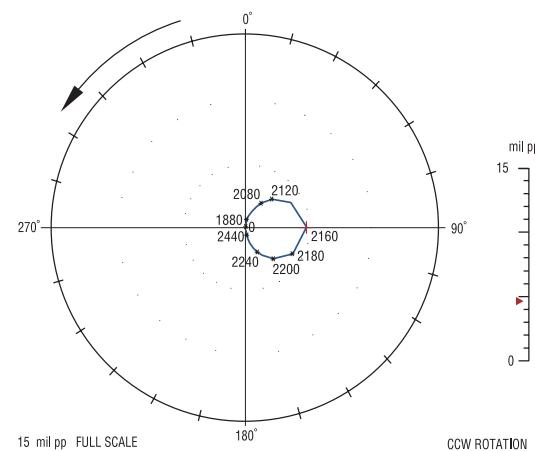


Figure 4. Polar plots showing the calculated residual unbalance response to new unbalance standard.

Balancing records from the high-speed balancing facility were obtained. The repaired and rebuilt rotor was balanced to a residual unbalance state of 24.1 g·m (33.5 in·oz). This value was used in the computer program for the forced response run.

The rotor mode shape at the balance resonance frequency is shown in Figure 2. This analysis offered clear evidence why the internal rub of the rotor had occurred. It was also clear that the allowable residual unbalance of the rotor currently under repair must be considerably more stringent.

Figure 3 shows the polar plots of the calculated rotor response at Bearing #1 and turbine midspan location. When the rotor speed was at the first balance resonance frequency, vibration at Bearing #1 was calculated to be 97.0  $\mu\text{m}$  (3.82 mil) pp. The midspan vibration was calculated to be 551  $\mu\text{m}$  (21.7 mil) pp.

Several additional computer response runs were made, with various weight distributions, to determine the maximum

allowable residual unbalance to ensure a successful re-commissioning of the turbine. The new standard was determined to be a maximum of 8.78 g·m (12.2 in·oz), about one-third of the original. The polar plots of the calculated response with this unbalance are shown in Figure 4.

This information was forwarded to the high-speed balancing facility. The turbine rotor was successfully balanced to this standard and returned to the job site for re-commissioning. The startup of the unit was successful and uneventful.

### Conclusion

The use of rotor dynamics modeling is a powerful tool to analyze the forced response of a rotor to unbalance forces and accurately determine the rotor mode shape. The model can be used to define the midspan deflection (vibration) of a rotor when field measurements are not possible. If this analysis had not been done and a new balance standard not established, the second startup of the turbine might have yielded the same midspan rubs. ☐

## ANNOUNCEMENT

### Special Year 2000 support line activated

**B**ently Nevada is dedicated to our Year 2000 compliance testing and customer support programs. While we have already run rigorous testing on all potentially affected products, and published that information on our website, we are still planning to be available to provide support on any compliant system issues.

Many of you have asked how we will handle any Year 2000 questions or problems during the holiday season, when the actual year rollover occurs, and whether we will have adequate support staff on duty to provide assistance. The answer is a resounding "yes." Our Technical Support Staff in Minden, Nevada, will be available for Year 2000-related issues, 24 hours a day, commencing at 12:00 am Pacific Time, Thursday, 30 December 1999 through 12:00 pm Pacific Time, Friday, 7 January 2000.

If you encounter, or think you might have, any Year 2000-related problems, please contact us. You may use our special 24-hour support line at 775-783-5055 (active only for the dates/times listed above).

For any general Year 2000 inquiries, please visit our website – [www.bently.com](http://www.bently.com). You'll find a wealth of online resources that will help you identify whether you are using any affected products, and how to make them Year 2000 compliant. ☐